TRANSIENT RESPONSE OF MARINE STEAM TURBINES WITH NON-INTERACTING CONTROLLERS

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ABSTRACT

A compound non-interacting regulating system for the bled steam pressure and the turbine's speed has been mathematically analyzed under sudden changes of external disturbances. The mathematical description of the control loops is analyzed in time domain, by digital computer simulation, in order to obtain the dynamic behavior of this complex control system with a wide variety of parametric analysis. The transients of the multi-variable control system with P,PD,PI or PID controllers are compared in what concerns their transient characteristics.

1.INTRODUCTION

Although the early development of steam power utilized the steam engine as a prime mover, the advantages of steam turbine regulation soon became apparent and have made it, the choice for all large modern steam propulsion plants [1]. Turbines are not size limited and can be provided for any power rating up to the maximum likely to be encountered in marine service. The marine steam turbine performance is mainly affected by the load disturbance which is interrelated to the propeller load. This factor may lead to change in propeller torque which will be considered to be an external load disturbance on the plant. Overall, the variation of the amount of extracted steam and consequently the variation of bled steam pressure will affect the turbine's speed too. This would represent another source of disturbance on the steam power plant.

In other words both variations of the amount of extracted steam and the torque disturbance will affect directly the dynamic behavior of the speed of the turbine. So, it is important to design a control system in order to maintain the transient response of the turbine's speed as small as possible.

This control system can be designed using two hydraulic servo-motors attached to a pressure controller and to a speed governor. The advantage of the hydraulic servo-motor is the very rapid action and it is capable of producing very large forces due to small displacement of input variable. Also, the pressure forces acting on it are balanced so that it requires little force to change its position.

Few previous work have dealt with the automatic extraction pressure,e.g. [1]. However, regulation for both speed and extracted steam pressure was presented as schematic model of control systems without the derivation of a mathematical model[2,3].

In[4] an excellent survey is made only for different regulating systems and schemes. Besides, some research workers are only concerning with the systematic structuring of steam turbines as control systems,e.g.[5].

The analysis of the control action for both turbine's speed and extracted steam pressure was achieved through simultaneously interacting controllers[6]. In despite of the less advantageous expected transient performance of regulating a steam turbine with non-interacting controllers- if compared to interacting controllers, the problems of simplicity, reliability and cost may be beneficial to be in concern. In this paper, a non-interacting regulation system for the bled steam pressure and the turbine's speed has been mathematically analyzed.

2.SYSTEM OPERATION

Figure (1) and (2) show typical control systems for compound regulation of bled steam pressure and turbine's speed with non-interacting controllers. Consider the system shown in Figure (1), the fly-weights are geared to the shaft, so that the rod displacement signal **W** may be changed relative to the output speed. The principle of operation of the system may be easily summarized as follows:

Suppose that the resisting torque on the propeller is increased which in turn will decrease the speed of the turbine. Due to the decreased centrifugal forces on the flyweights, the pilot valve is displaced downwards which allows pressurized oil to move the power piston upwards increasing the amount of steam to the main governing valve. Hence, the speed of the turbine starts to increase. According to the principle of automatic feed back loop, the actual output is always measured, feedback and compared to the desired one. A correcting error signal puts the controller into action either directly or indirectly - through actuators- on the control plant.

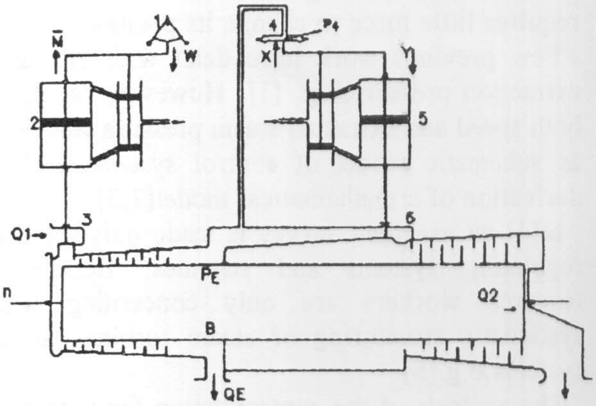

Figure 1.

Now suppose that the steam demand Q_E increases, the pressure P_E in the extraction chamber will decrease, consequently the pressure P_4 inside the pressure sensor decreases. This will cause the hydraulic servo-motor to move closing partially the pressure control valve. Since, the amount of the admitted steam to the turbine decreases, the turbine's power and consequently its speed will decrease. This will be sensed by the flyweights causing the hydraulic servo-motor to move upwards opening partially the speed control valve (main governing valve). This increases the steam amount admitted to the first portion of turbine's stages, and consequently increases the power and the speed, and vice versa.

The control concept of Figure(2) is identical to that of Figure (1) except that the regulating techniques are different; whereas the principle of hydromechanical control is used in Figure (1), the flapper- nozzle principle is adopted to the scheme shown in Figure(2).

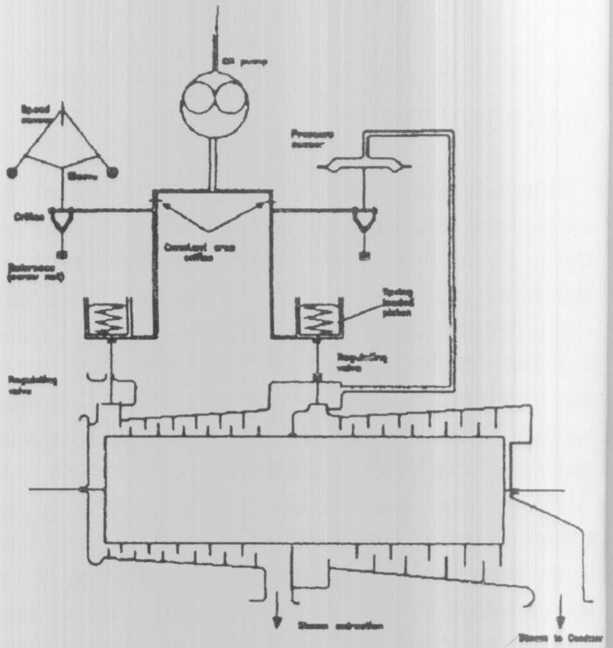

Figure 2.

3.MATHEMATICAL MODELLING

The overall block diagram of the control system is shown in Figure (3), indicating the transfer functions and control properties for each control element illustrated as a unit step response to a unit input signal.

The pressure governor has P_o control action while $(PID)_o$ speed governor is used. Both controllers are ideal, i.e. without time delays. Also, energy hydraulic servo-motors, with P_1 control property are attached to both controllers.

In order to build the mathematical model for each control component, references from [7] to [12] may be referred to. By the use of the superposition principle, the block diagram reduction yields the relative speed deviation n(s) in terms of the relative load Z(s) and the relative steam extraction $Q_E(s)$ disturbances, which may be written in a matrix form as:

$$n(s) = \frac{1}{F_1}\left[a_{11}\ a_{12}\right]\cdot\begin{bmatrix} Q_E(s) \\ Z(s) \end{bmatrix}$$

where

$$F_1 = (1+G_2\ G_3)(1+G_1\ G_5G_6\ G_7)+G_2G_3G_4G_5G_6G_7,$$

$$a_{11} = -\ G_2\ G_3\ G_4\ G_6\ ,$$

$$a_{12} = -\ G_6\ (1+G_2\ G_3)\ ,$$

and the values of the transfer functions from G_1 through G_7 are indicated in Figure (3).

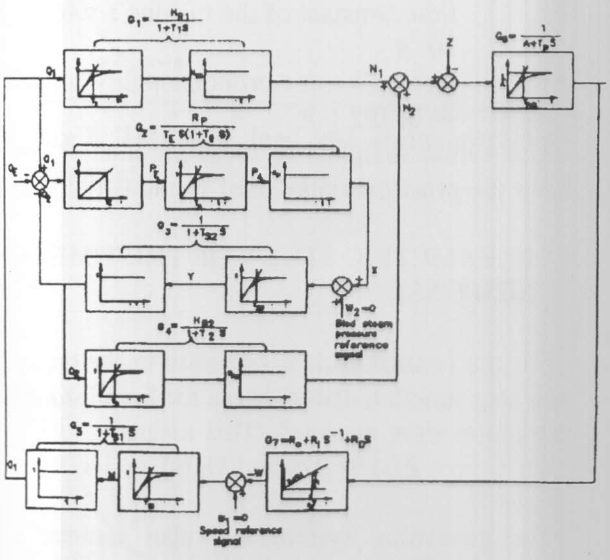

Figure 3.

Moreover Figures (4),(5) and (6) illustrate the modelling of this dynamic system by state variable signal flow graph, state variable diagram and analog simulation respectively. For the determination of the values of the parameters of $\bar{A}$, B,.., M and N the reader may refer to [12].

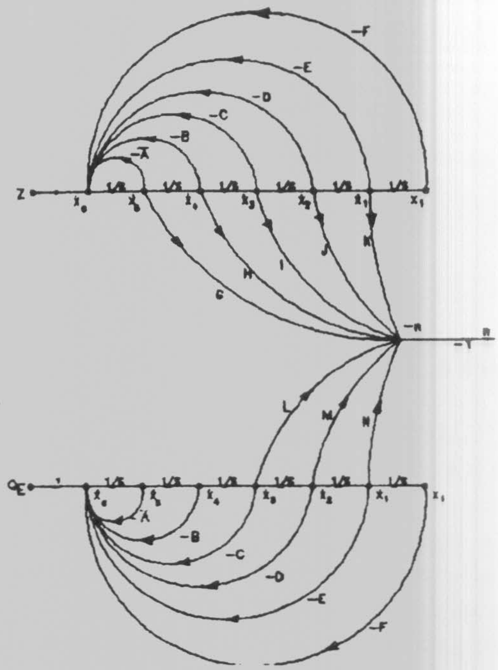

Figure 4.

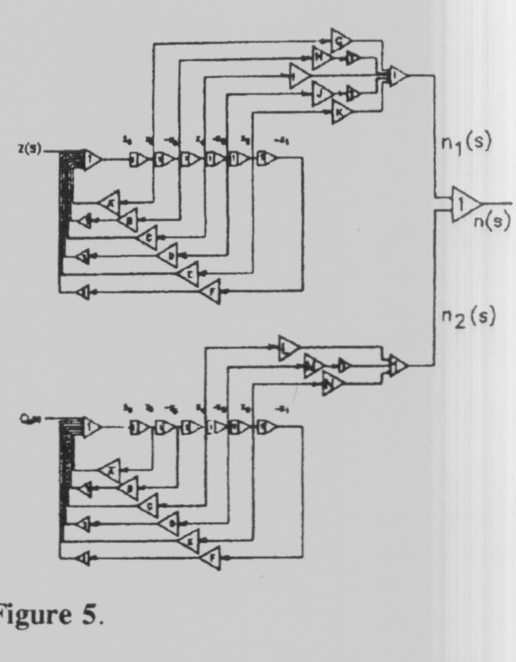

Figure 5.

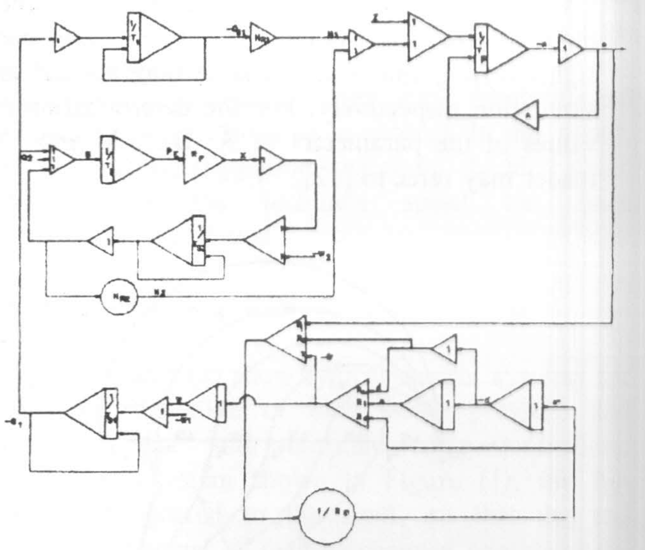

Figure 6.

4 COMPUTER-AIDED TIME - DOMAIN ANALYSIS OF CONTROL LOOPS:

The digital computer simulation for the analysis of the scheme shown in Figure (1) and whose block diagram is illustrated in Figure (3) was carried out to investigate the absolute stability, relative stability and the transient response under both external load and extracted steam amount disturbances. The main problem is to compute the roots of the characteristic polynomial, which are necessary for several reasons, especially to study the stability of the system. Having extracted the roots, the partial fraction parameters can be calculated, which will form the inverse laplace transform parameters. Finally, a plot of the time function may be produced, using the inverse transform and employing a technique for picking up the time increments for plotting. This was achieved by three FORTRAN programs run at the computer center of the faculty of engineering,Alexandria University.

5.NOMENCLATURE AND DATA:

In order to study the dynamic behavior of the system shown in Figure (1), scanned data range for the different system parameters can be considered. The parameters values are stated clearly in Ref.[12], and will be given as:

H_{R1}, H_{R2} ratio of heat drop before and after steam extraction related to the total heat drop

$$H_{R1} = 1/3 \ (-), \quad H_{R2} = 2/3 \ (-),$$

R_o gain of the speed governor = 20 and 25 (-),

R_I integral property coeff. of the speed governor = 0, 0.5 and 0.8 s^{-1},

R_D derivative property coeff. of the speed governor = 0, 5 and 8 (-),

R_P pressure governor's gain = 20 (-)

T_1, T_2 time delays due to steam accumulators after the main and second steam governing valves respectively

$$T_1 = 0.2 \text{ s and } T_2 = 0 \text{ s}$$

T_E= (extraction room vol.* max. extracted steam press.)/(superheated steam const.* abs. steam temp.* max. mass rate of steam extraction)

$$T_E = 10 \text{ s}$$

T_g time delay of the pressure controller= 0 s

T_{s1}, T_{s2} time constants of hydraulic servo-motors of the main and second governing valves respectively

$$T_{s1} = 0.1 \ \& \ 0.3 \text{ s}, \quad T_{s2} = 0.2 \text{ s}$$

T_P time constant of the turbine's rotor = 16 and 20 s

A turbine's rotor proportionality constant = 0, 0.2 (-).

The values assigned to these parameters are picked from the practical range used in industrial technology

6. PARAMETRIC STUDY OF THE TIME DOMAIN RESPONSE:

For the control system response in the time domain unit step function disturbances for both load and steam extraction were assumed. This means:

$$Z(s) = 1/s \text{ and } Q_E(s) = 1/s.$$

The governing system was also assumed to have symmetrical dynamic behavior in cases of either the opening or the closure of the control valves. Figures from (7) to (15) inclusive show the transients of percentage speed deviation with respect to time for the control system under consideration subjected to unit step load and steam extraction disturbances through the prementioned scanned data range. The control system insures an acceptable maximum speed deviation (maxim overshoot) not exceeding 10% with the majority of the combinations of the plant's dynamics and the regulating system characteristics.

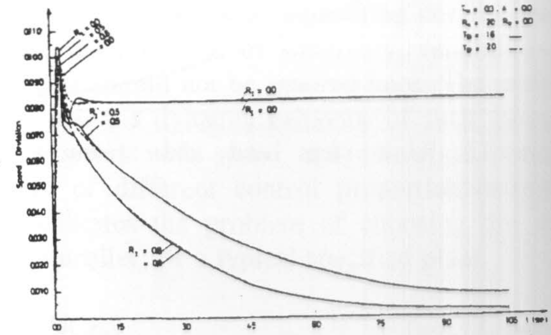

Figure 7. Unit step load and steam extraction disturbances.

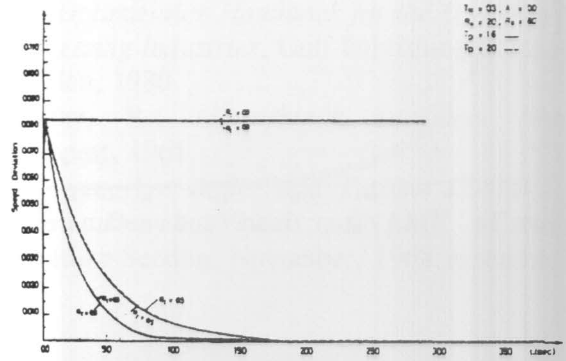

Figure 8. Unit step load and steam extraction disturbances.

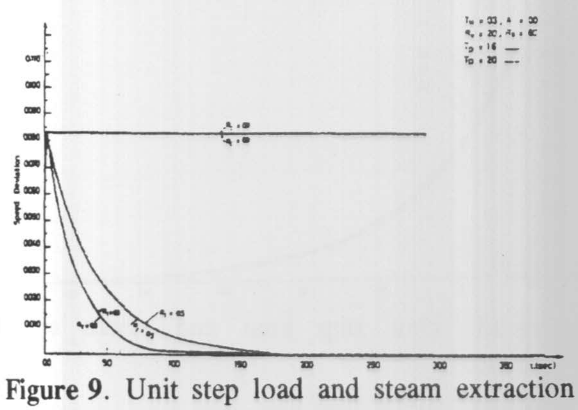

Figure 9. Unit step load and steam extraction disturbances.

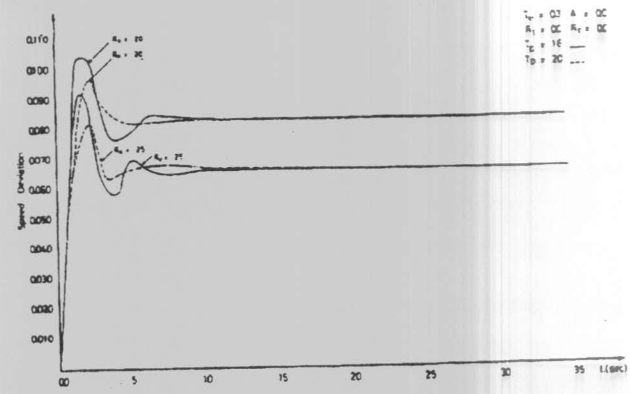

Figure 10. Unit step load and steam extraction disturbances.

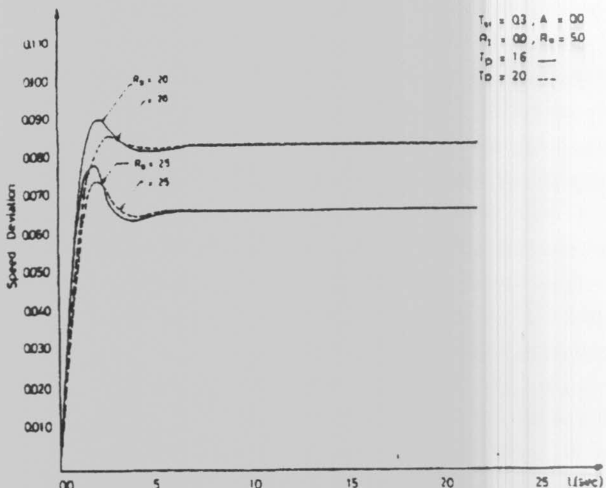

Figure 11. Unit step load and steam extraction disturbances.

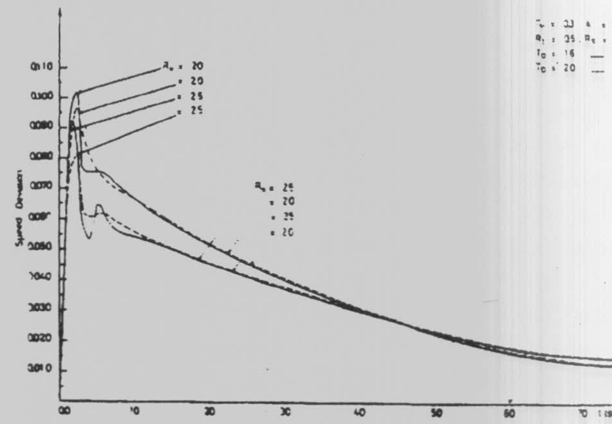

Figure 12. Unit step load and steam extraction disturbances.

Figure (7) shows a comparison of the transient response under the control of P and PI controllers. It may be seen that the P controller reaches the steady state swifter than the PI controllers. Furthermore, the P controller gives a higher speed deviation with larger peak time if compared to PI controllers. In other words, increasing R_I parameter may decrease the speed deviation for the same values of R_o and R_D. Figure (8) illustrates the dynamic behavior of the control system using PD and PID controllers. Zero speed deviation can be achieved by means of PID controllers. Whereas, higher maximum speed deviation can be represented by the use of PD controllers. Figure (9) illustrates the transient response characteristics of the control system for P and PD controllers. From these curves, it can be seen that with $R_o = 20$, increasing R_D from 0 to 8 (s) may result in a reduction in the maximum speed deviation with a smaller settling time. By selecting P controllers only with different gains, increasing of the controllers gain can decrease the maximum speed deviation as shown in Figure (10). For PD controllers, a significant difference in the maximum over shoot can be seen in Figure (11). Furthermore, with R_D = 5 (s), increasing of R_o from 20 to 25 may give a reduction of about 18% in the maximum speed deviation.

Figure (12) represents the transient response of speed deviation by using PI controllers with different values of R_o , for the same integral parameter $R_I = 0.5\ s^{-1}$.

Figure (13) illustrates the transients of the multi variable control system either with PI or PID controllers. Whereas Figure (14) shows the effect of varying the controllers gain R_o on the transients with PID controllers. The results indicate a reduction in the maximum overshoot approaching 1.5% when increasing R_o from 20 to 25. While it depends on which value is specified for the settling time ($\pm$ 2% or $\pm$ 5%) that the control designer may select a 20 or 25 as the controllers gain.

Varying the value of the turbine's rotor proportionality constant A= 0 (specifying an integral "I_o" rotor) to A = 0.2 (specifying a proportional "P_1" rotor) has a negligible effect on the transient response Figure (15). Finally it must be noted that the effect of changing the time constant of the turbine's rotor from 16 to 20 (s) does not have a considerable effect either on the maximum speed deviation, on the peak time, or on the settling time for the selected values of the other parameters.

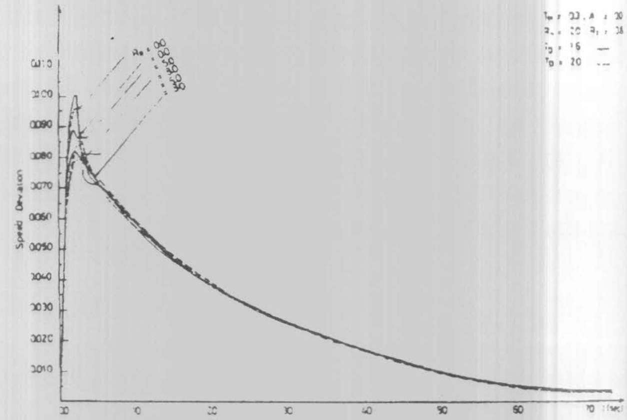

Figure 13. Unit step load and steam extraction disturbances.

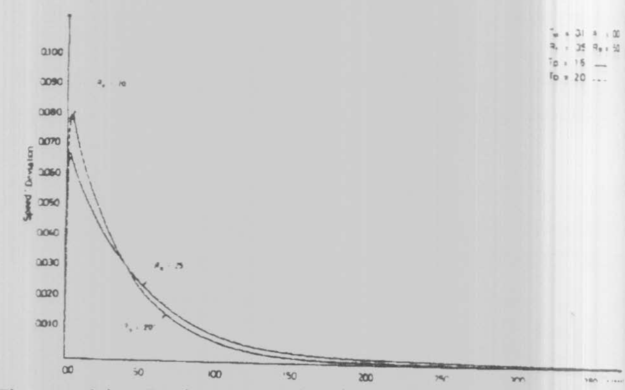

Figure 14. Unit step load and steam extraction disturbances.

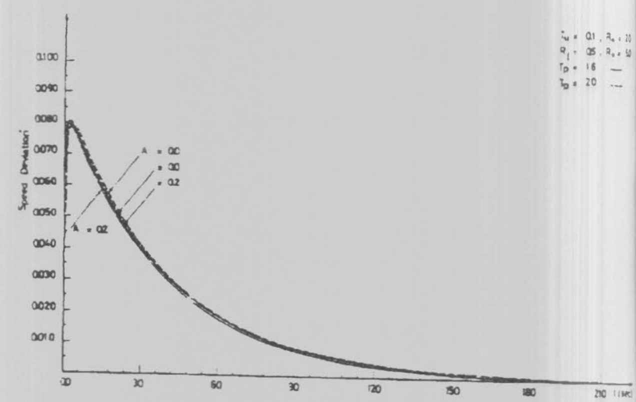

Figure 15. Unit step load and steam extraction disturbances.

CONCLUSION

The transient response of marine steam turbines with non-interacting controllers have been investigated. A wide variety of the plant's ,as well as, the controller's properties and parameters was dealt with. The research reveals also the slower response with relatively poor time domain characteristics if compared to such multi variable control loops with interacting controllers[6]. However, the problems of simplicity, reliability and relative cost should not be ignored here. The analysis displays also the dynamic behavior of such complex control systems with speed and extraction pressure regulators of different control properties; a matter which facilitates the problem of choosing the most proper controller for a typical specified plant.

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